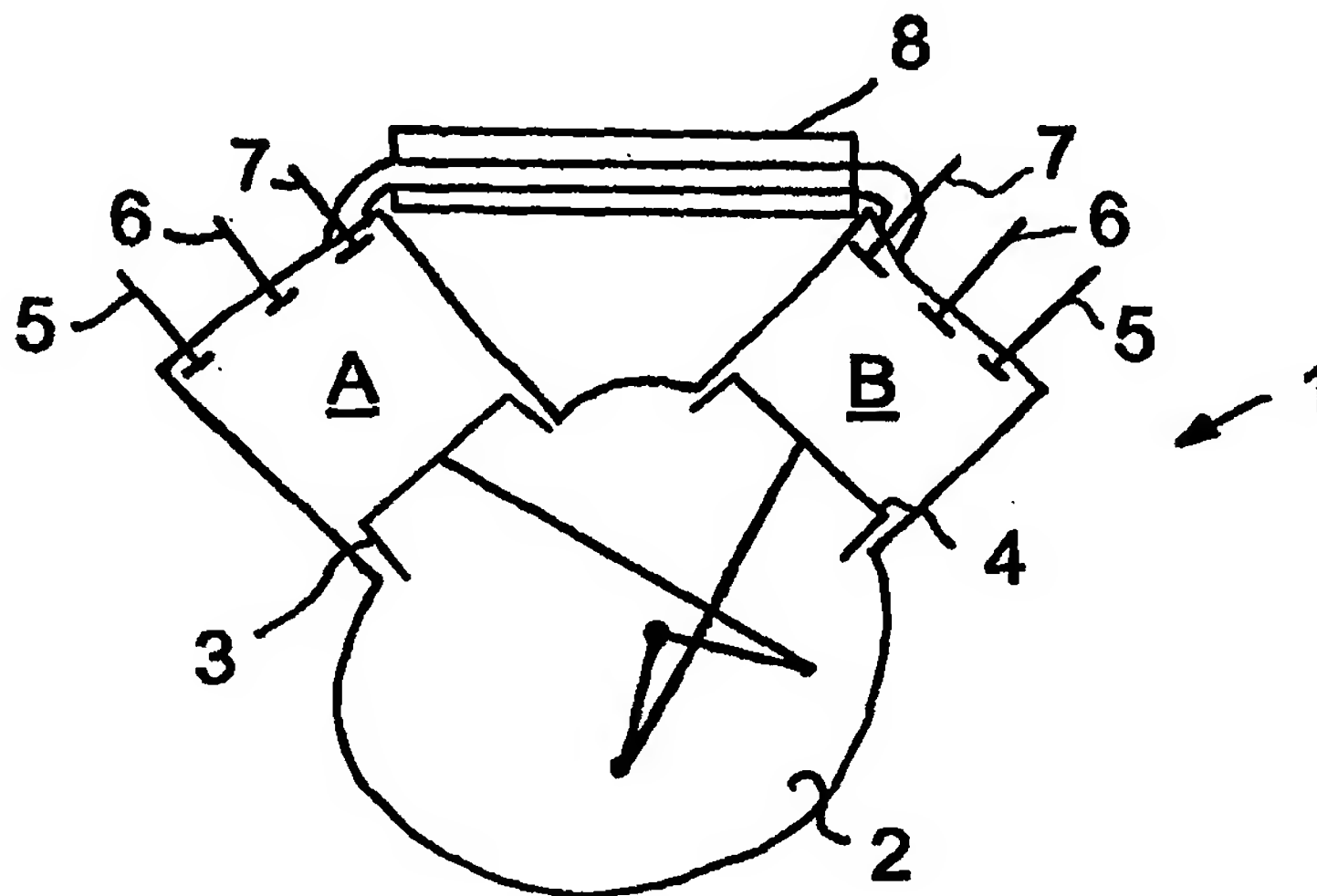




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(21) International Application Number: PCT/SE98/02335 (22) International Filing Date: 16 December 1998 (16.12.98) (30) Priority Data: 9704704-7 17 December 1997 (17.12.97) SE (71) Applicant (for all designated States except US): SCANIA CV AKTIEBOLAG (publ) [SE/SE]; S-151 87 Södertälje (SE). (72) Inventor; and (75) Inventor/Applicant (for US only): LUNDQVIST, Ulf [SE/SE]; Svartviksslingan 27, 8 tr, S-167 38 Bromma (SE). (74) Agent: WALDEBÄCK, Hans; Scania CV AB, Patents, S-151 87 Södertälje (SE).		(81) Designated States: BR, DE, JP, US, European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE). Published <i>With international search report.</i> <i>In English translation (filed in Swedish).</i>
(54) Title: ARRANGEMENT FOR RECIRCULATION OF EXHAUST GASES IN A COMBUSTION ENGINE WITH AT LEAST TWO CYLINDERS (57) Abstract <p>An arrangement for exhaust gas feed-back in a four-stroke combustion engine with at least two cylinders (A, B) cooperating with one another, whereby each cylinder in the engine is connected by a transfer line (8) to another cylinder operating with substantially approximately 360° displacement. Exhaust gas feedback is made possible by using the over-pressure prevailing in the respective cylinder (A, B) during its expansion stroke to feed exhaust gases back to the cooperating cylinder during the latter's induction stroke. This is made possible by the fact that the transfer line is provided with controllable valves (8) at both of its respective ends and by the fact that the transfer line leads into the respective cylinders.</p>		



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Arrangement for recirculation of exhaust gases in a combustion engine with at least two cylinders

The present invention relates to an arrangement in accordance with the preamble to patent 5 claim 1.

STATE OF THE ART

A known practice in combustion engines is to feed exhaust gases back to the engine inlet
10 in order to reduce the nitrogen oxides content of the exhaust. The exhaust gases fed back have the effect of lowering the combustion temperature and hence reducing the proportion of the sulphur in the inlet air which can be converted to nitrogen oxides. In Otto engines, this technique, usually called EGR (exhaust gas recirculation), has become widely used as a relatively simple way of reducing the content of harmful exhaust
15 emissions. In diesel engines, however, this technique has not become so widely used, because diesel engines have particular problems which make it impossible to apply directly to them solutions which apply to Otto engines.

One of these particular problems is that combustion in diesel engines normally takes place
20 with excess air. This results indirectly in relatively large volumes of exhaust gases having to be transferred for a relatively large proportion of engine operating time if the intended function is to be achieved. This problem is exacerbated in the case of engines of the supercharged type in that the pressure in their inlet system is greater than the pressure in their exhaust system for a large proportion of their operating time. As the majority of
25 modern diesel engines are supercharged, this is an obvious problem.

Among known solutions usable in supercharged engines, a distinction may be made between two main principles usually referred to respectively as "short route EGR" and "long route EGR". In short route EGR, the exhaust gases are taken from a point upstream
30 of an exhaust turbine arranged in the exhaust system and are fed back at a point downstream of an inlet air compressor arranged in the inlet system. In long route EGR, the exhaust gases are taken from a point downstream of the exhaust turbine and are fed back at a point upstream of the inlet air compressor. Both principles have advantages and

disadvantages. An advantage of the short route solution is the possibility of preventing fouling of a charge air cooler for the inlet air. A disadvantage of the short route solution is that it requires some form of pressure increasing arrangement because the exhaust gases are fed back at a point where the pressure is normally higher than the pressure prevailing
5 in the exhaust manifold of the exhaust system.

A known practice is to use a separate supercharging unit to bring about this increase in the pressure of the exhaust gases, as described, for example, in WO96/18030 and WO96/18031. A disadvantage of those solutions is the need for the extra supercharging
10 unit or some other pressure increasing device, which makes those solutions both expensive and bulky.

A known technique with regard to the present invention which may also be mentioned is US 4 422 430, which refers to exhaust gases being fed back to the inlet of a four-stroke
15 Otto engine with a view to improving the mixture of air and fuel which is fed into a cylinder. In that case, exhaust gases are fed back from a cylinder via a direct line to the inlet of another cylinder whose working cycle is displaced 360 crankshaft degrees. Exhaust gases are thus led from a cylinder which is in its expansion stroke to the inlet of a cylinder which is in its induction stroke. The exhaust gases thus fed back cause in the
20 inlet air concerned a turbulence which promotes effective mixing of fuel and air, while at the same time the hot exhaust gases fed back are intended to further improve the vaporisation of the fuel. Even if the object is not to feed exhaust gases back for the purpose indicated above, this solution does cause a feedback of exhaust gases. This feedback does not take place, however, in any controllable manner and even in this case
25 there are difficulties in feeding the exhaust gases back in a supercharged engine, since they are fed back to the inlet ducts of the engine.

OBJECT OF THE INVENTION

30 An object of the present invention is to eliminate the problems of the state of the art when using a solution for exhaust gas feedback according to the short route alternative in a supercharged diesel engine. The invention thus aims to make effective exhaust gas

feedback possible without using a separate supercharging unit or other pressure increasing device for the exhaust gases fed back.

According to the invention, this is achieved in an arrangement of the kind mentioned in 5 the introduction by its being provided with the features indicated in the characterising part of patent claim 1.

The invention makes it possible for the pressure which normally prevails in a cylinder during the expansion stroke to be used for transferring exhaust gases to a cylinder which, 10 owing to supercharging, has a relatively high inlet pressure. The need to provide a charging compressor or the like for feeding the EGR gases in is thus eliminated, thereby simplifying and reducing the cost of the installation.

The fact that the exhaust gases fed back go directly to the cylinders and not to the inlet 15 ducts means that the engine can be supplied with the amount of air required for optimum combustion. The exhaust gases fed back do not reduce the amount of air entering the cylinders of the engine.

Arranging valves in both ends of the transfer line makes it possible for exhaust gas feedback to take place independently of the engine's ordinary valves and to optimise both 20 the amount and the timing of the feedback.

An advantageous embodiment of the invention makes it possible for cooling of the exhaust gases fed back to take place in the transfer line, which thus also serves as an effective cooler, thereby eliminating the need for special cooling measures. 25

Suitable dimensioning of the volume of the transfer line makes it possible to further improve the cooling of the exhaust gases.

Further features and advantages distinguishing the invention are indicated in the 30 embodiments described below.

DESCRIPTION OF THE DRAWINGS

Two embodiments exemplifying the invention will now be described in more detail with reference to the attached drawings, in which:

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Fig. 1 depicts schematically a first embodiment of the invention in a four-stroke diesel engine of V8 type,

Fig. 2 depicts schematically the interlinking of the cylinders in an engine according to Fig. 1,

10 Fig. 3 depicts schematically a second embodiment of the invention in a six-cylinder in-line engine,

Fig. 4 depicts a cross-section through a transfer line,

Fig. 5 depicts part of a pressure/volume diagram for a diesel engine to illustrate the function of the invention.

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DESCRIPTION OF EMBODIMENTS

Fig. 1 depicts schematically a combustion engine 1 of piston engine type with eight cylinders which are arranged in two banks set at an angle to one another, i.e. it is a type of engine usually referred to as a V8 engine. It is a four-stroke diesel engine intended, for example, for a heavy-duty vehicle such as a truck or a bus. Fig. 1 depicts schematically two cylinders A and B whose working cycles are displaced 360 crankshaft degrees. Each cylinder is provided in a conventional manner in its cylinder head with at least one inlet valve 5 for supply of combustion air and at least one exhaust valve 6 for removal of exhaust gases arising from the combustion. The inlet valves 5 and the exhaust valves 6 are controlled in a conventional manner by an undepicted camshaft.

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Also arranged in the cylinder head and between the two cylinders A and B in accordance with the present invention is a transfer line 8 designed to transfer EGR gases between the cylinders A and B. The transfer line 8 leads into the two cylinders A, B and is provided at each end with a valve 7 by means of which the connection between the respective cylinders and the transfer line 8 can be opened and closed. The transfer line 8 leads into the cylinder head of the engine, to make it easy to carry out the machining of the relevant

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ducts and the fitting of the valves. This also makes it easier to provide cooling of the ducts. In this example, the valves 7 take the form of mushroom valves of the same type as conventionally used for inlet and exhaust valves of combustion engines. These valves 7 are controllable separately and hydraulically by a separate control system not depicted 5 in more detail. In alternative embodiments, the valves 7 may instead be controlled by camshafts, by electrical means or in another manner known per se.

Fig. 2 shows how the other cylinders of the engine are connected together in pairs in a similar manner to those depicted in Fig. 1. Each cylinder C1-C4 in one bank of cylinders 10 is thus connected to a cylinder C5-C8 in the other bank of cylinders. The cylinders A and B depicted in Fig. 1 thus correspond to each pair of cylinders connected to one another according to Fig. 2. The paired cylinders connected to one another have working cycles displaced 360 crankshaft degrees relative to one another. It may be noted that all four strokes of the engine extend over 720 crankshaft degrees, which is equal to two whole 15 crankshaft turns. In this description, the four cycles are referred to respectively as the induction stroke, the compression stroke, the expansion stroke and the exhaust stroke. Each cylinder which is in an expansion stroke will thus be connected to another cylinder which is in the induction stroke, and so on.

20 Figure 4 shows a cross-section through one of the transfer lines 8 between the paired cylinders C1-C8. The transfer line 8 consists of an outer casing 8" which contains a number of internal ducts 8' (in this example six internal ducts 8') which form the actual transfer lines for transfer of exhaust gases between the two cylinders A,B connected to one another. The outer casing delineates a cooling duct through which passes a cooling 25 medium which cools the internal ducts 8' and hence also cools the exhaust gases in the internal ducts 8'. It is advantageous to use the ordinary cooling medium of the engine, in which case the lines concerned form part of the engine's ordinary cooling system. The internal ducts 8' are connected to one another at the respective ends of the transfer line to form a common duct, the two ends of which can be closed by the respective valves 7.

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In this embodiment the respective transfer lines 8 are depicted in the form of a separate line. In practice they may take the form of ducts cast or drilled in the cylinder head or other parts of the engine, in which case the internal ducts 8' take the form of a number of

tubes (e.g. copper tubes laid in parallel and accommodated in the ducts), which ducts have the engine's ordinary cooling fluid passing through them. The transfer lines 8 thus constitute a cooler for the exhaust gases fed back. In other alternative embodiments the cooling may be by a separate cooling system. The transfer line 8 is provided with
5 arrangements for cooling along substantially the whole of its length in order to bring about as effective cooling as possible.

Another aspect of the transfer lines 8 which is important, at least in an advantageous embodiment of the invention, is that the combined volume of the internal ducts 8'
10 accommodating the exhaust gases for the respective lines 8 is substantially the same for all the lines 8. These volumes in the respective lines are themselves equal to the amount of exhaust gases normally required to be transferred during one of the cycles of the four-stroke engine. For an engine in accordance with this embodiment, the volume of the respective transfer lines is approximately one-third the volume of the respective cylinders.
15 For this reason all the transfer lines have the same diameter and the same length, which means that they accommodate the same volume. The reason for this is indicated further on in this description.

Fig. 3 shows an example of the interlinking by transfer lines 8 of cylinders C1-C6 in pairs
20 in an engine of the "straight six" type, whereby cylinders 360 degrees apart are connected together. In accordance with the foregoing, here again all the transfer lines are of the same length and accommodate equal volumes.

In both of the alternatives referred to above, which cylinders are interlinked with one
25 another depends on the ignition sequence of the engine, and engines with different ignition sequences will have different cylinders interlinked with one another.

The functioning of the arrangement described above is illustrated by Fig. 5, which shows the lower part of a pressure-volume (p-v) diagram for a diesel engine. The y-axis in the
30 diagram represents the pressure P in one of the cylinders of the engine and the x-axis the momentary cylinder volume V. The curve depicted represents, in a conventional manner, the four strokes or phases of the engine, i.e. the induction stroke 13, the compression stroke 11, the expansion stroke 10 and the exhaust stroke 12. For the purposes of the

description below, the cylinder concerned is assumed to correspond to cylinder A, which in accordance with the foregoing is connected to and cooperates with cylinder B via the transfer line 8.

5 It is initially supposed that in cylinder A combustion has recently taken place and the combustion gases are expanding, which corresponds to a working point moving downwards along the curve section 10. The inlet and exhaust valves 5, 6 of cylinder A are supposed to be closed, as also the two valves 7 at the ends of the relevant transfer line 8. At the end of the expansion stroke 10, corresponding to a point 9 on the curve, the
10 valve 7 opens to the transfer line 8 in the relevant cylinder A and at approximately the same time the exhaust valve 6 opens so that the exhaust gases can be led further into the exhaust system, conventionally first to a turbine in a turbocharger and then to a silencer. As the valve 7 has opened to the transfer line 8, exhaust gases flow into the transfer line 8 and their high pressure makes their inflow resemble a pulse wave or a pressure wave.

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While this was taking place, the cooperating cylinder B was in its induction stroke 13. At approximately the time when the exhaust valve in cylinder A opened, corresponding to the point 9, the inlet valve into the second cylinder B had recently closed, corresponding to point 14 on the curve. Slightly after the inlet valve into cylinder B closed, the transfer
20 line valve 7 of cylinder B opened. This stage, represented by the point 15, is only slightly later than the time when the transfer line valve 7 of cylinder A opens. Consequently, the exhaust gases in the transfer line 8 flow into cylinder B. The time between the two valves 7 of the transfer line opening and the length of the transfer line is adjusted so that the transfer can take place with utilisation of the pressure wave which causes the flow of
25 exhaust gases into the transfer line 8 from cylinder A.

At the end of the expansion stroke 10, the exhaust valve in cylinder A will close and its inlet valve will open, advantageously with a certain overlap, and cylinder A will change over to an exhaust stroke 12. This is preceded by the transfer line valve 7 of cylinder A
30 closing, represented by point 16. During this time, cylinder B will have changed over to the compression stroke 11. Despite the engine's crankshaft turning through a relatively large number of degrees, the pressure will not have risen appreciably in cylinder B. The diagram shows at 16 the point at which the transfer line valve 7 of cylinder A closes and

at 17 the time when the transfer line valve 7 of cylinder B closes. The diagram shows that the pressure at point 16 is higher than at point 17, thereby always ensuring that exhaust gases can be fed back by their own pressure.

- 5 The opening of the exhaust valve 6 and the opening of the valve 7 at the end of the transfer line, corresponding to point 9, take place at a pressure p_1 , which in a conventional modern engine corresponds to a pressure slightly higher than a 9 bar absolute pressure at full load. This pressure p_1 exceeds the pressure p_2 prevailing in the cooperating cylinder B, which at this stage is at the end of the induction stroke 13 and is at a pressure of
- 10 approximately 3 bar absolute in a modern supercharged diesel engine. The pressure difference between p_1 and p_2 thus makes it possible, with a good margin, for the exhaust gas content of the transfer line 8 to be transferred to the receiving cylinder B.

The exhaust gases which flow into the transfer line from cylinder A are relatively hot,

15 whereas the exhaust gases hitherto in the transfer line 8 will have cooled. Relatively cool exhaust gases are thus introduced into cylinder B.

It is also possible to realise from the diagram that the pressure during the exhaust stroke 12 is lower than the pressure during the induction stroke 13, this being a consequence of

20 the engine being supercharged. Hence the problem described in the introduction that the exhaust gases cannot be transferred to the inlet without being subjected to a pressure increase.

Appropriate volume adaptation in the transfer line 8 ensures that substantially only the

25 properly cooled exhaust gases situated in the transfer line are transferred, which leads to these cooled gases being replaced in the transfer line by the hot gases from the delivering cylinder, which gases will then remain in the transfer line during a subsequent crankshaft turn for them to cool down and thereafter be fed back in the next cycle accordingly.

- 30 When cylinder B has completed the compression stroke 11, it changes over, after the combustion, to the expansion stroke 10. In a similar manner to that described above for cylinder A, cylinder B will now follow the same pattern, but with the difference that it

will now be the hot exhaust gases from cylinder B that are led into the transfer line 8, while the cooled exhaust gases in the transfer line 8 are fed back to cylinder A.

To make it possible to adjust the amount of exhaust gases fed back to the cylinders, the 5 valves 7 of the transfer lines are separately controlled. In situations requiring more exhaust gases to be fed back the valves 7 are controlled so that they open earlier, and in situations which require smaller amounts of exhaust gases to be fed back the valve 7 of the delivering cylinder opens later, all this relative to the situation described above in which these valves 7 open approximately simultaneously with the exhaust valve.

10 However, the transfer line valve 7 of the delivering cylinder is nevertheless controlled so as to close at approximately the same time irrespective of the amount of exhaust gases transferred.

The control of the valves 7 and hence the control of the amount of exhaust gases to be fed back by means of the undepicted control system, which in a conventional manner 15 incorporates sensors for detecting engine parameters appropriate to the purpose.

A very substantial aspect in this context is that the EGR gases transferred are properly cooled, preferably to such an extent that the water vapour contained in them condenses to liquid water. This has a particularly good effect on the combustion process in the 20 cylinder so that the temperature increase is reduced. It also has a certain cleaning effect on valves and associated seats, thereby counteracting sticking.

A substantial benefit of the invention is that although the amount of EGR gases drawn off certainly results in lower power of the engine's turbine, this power is recovered by the 25 EGR gases being fed directly into the cylinder without detours via turbines and compressors. In aggregate there is therefore no appreciable loss of charging pressure.

The invention works in all four-stroke engines (not only diesel engines) with paired cylinders displaced approximately 360 crankshaft degrees, but may also be made to work 30 in such cases as, for example, five or seven cylinders if appropriate delays are ensured by correct line dimensioning.

The invention is particularly advantageous in supercharged diesel engines but may with advantage also be used both in unsupercharged engines and in other types of engines.

PATENT CLAIMS

1. Arrangement for exhaust gas feedback in a four-stroke combustion engine with at least two cylinders (A,B) each incorporating at least one inlet valve (5) and at least
5 one outlet valve (6), whereby each cylinder is arranged to cooperate with another cylinder operating with substantially 360 degree displacement, and whereby a transfer line (8) for exhaust gas feedback is arranged between cylinders cooperating in pairs, **characterised** in
that the transfer lines (8) lead into the cylinder space of respective cylinders (A,B),
10 that the transfer lines (8) are provided at their respective ends with valves (7), and that the valves (7) are controlled so that the pressure in the one cylinder during its expansion stroke (10) is utilised for exhaust gas transfer to the other cylinder.
2. Arrangement according to claim 1, **characterised** in that the valves (7) of
15 the transfer lines are separately controllable, thereby making exhaust gas feedback between the cylinders (A,B) possible independently of the inlet and outlet valves (5,6).
3. Arrangement according to claim 1, **characterised** in that the transfer lines
8[(8)] lead into the cylinder head of the respective cylinders.
20
4. Arrangement according to claim 1, **characterised** in that each transfer line
(8) has a volume which substantially corresponds to the volume of the amount of exhaust gases which is fed back to a cylinder during one of the working cycles of the four-stroke engine so that, during operation, it is essentially only combustion gases which have
25 remained in the transfer line (8) during a crankshaft turn that are fed into the cylinder (A,B) which receives the exhaust gases.
5. Arrangement according to claim 4, **characterised** in that each transfer line
(8) has a volume equal to approximately one-third of the volume of the respective
30 cylinder.
6. Arrangement according to claim 1, **characterised** in that each transfer line
(8) is cooled along substantially the whole of its length.

7. Arrangement according to claim 6, **characterised** in that each transfer line (8) along substantially the whole of its length takes the form of a cooler with a multiplicity of parallel tubes accommodated in a cooling duct, and that the ordinary 5 cooling medium of the engine is used for cooling the transfer lines (8).

8. Arrangement according to claim 1, **characterised** in that the transfer line valve (7) in the cylinder which receives exhaust gases from the transfer line (8) is arranged to open after the cylinder's inlet valve (5) has closed.

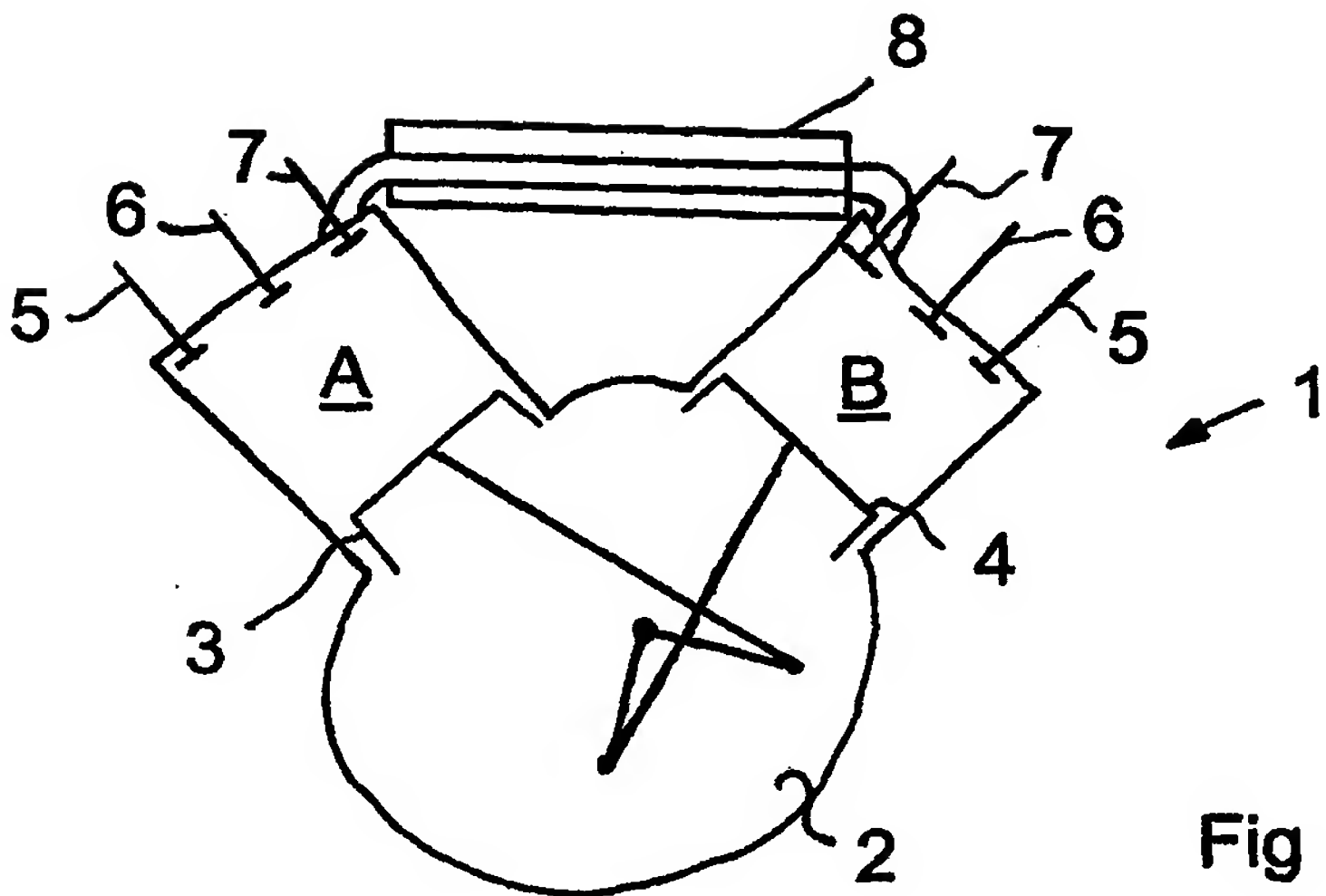


Fig 1

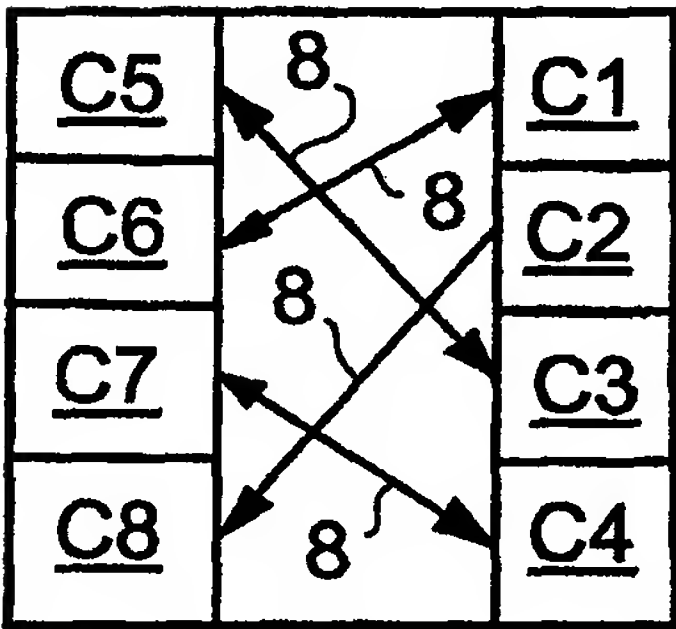


Fig 2

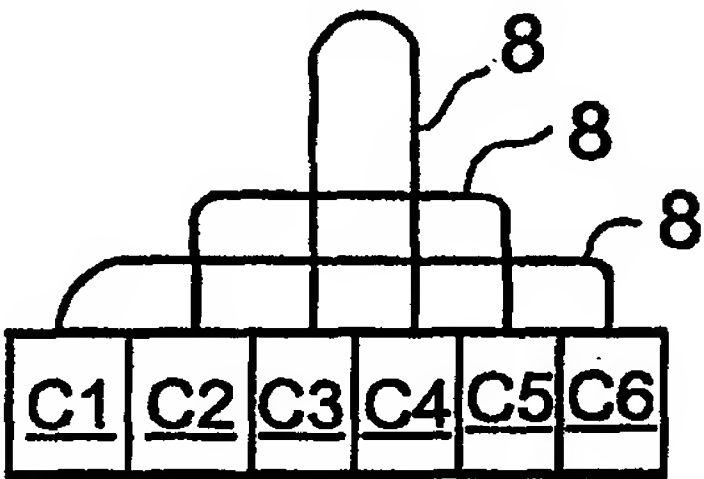


Fig 3

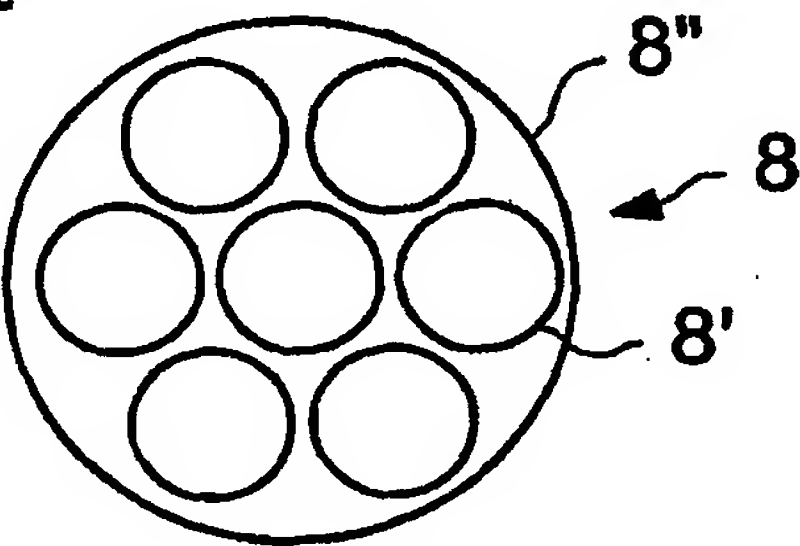


Fig 4

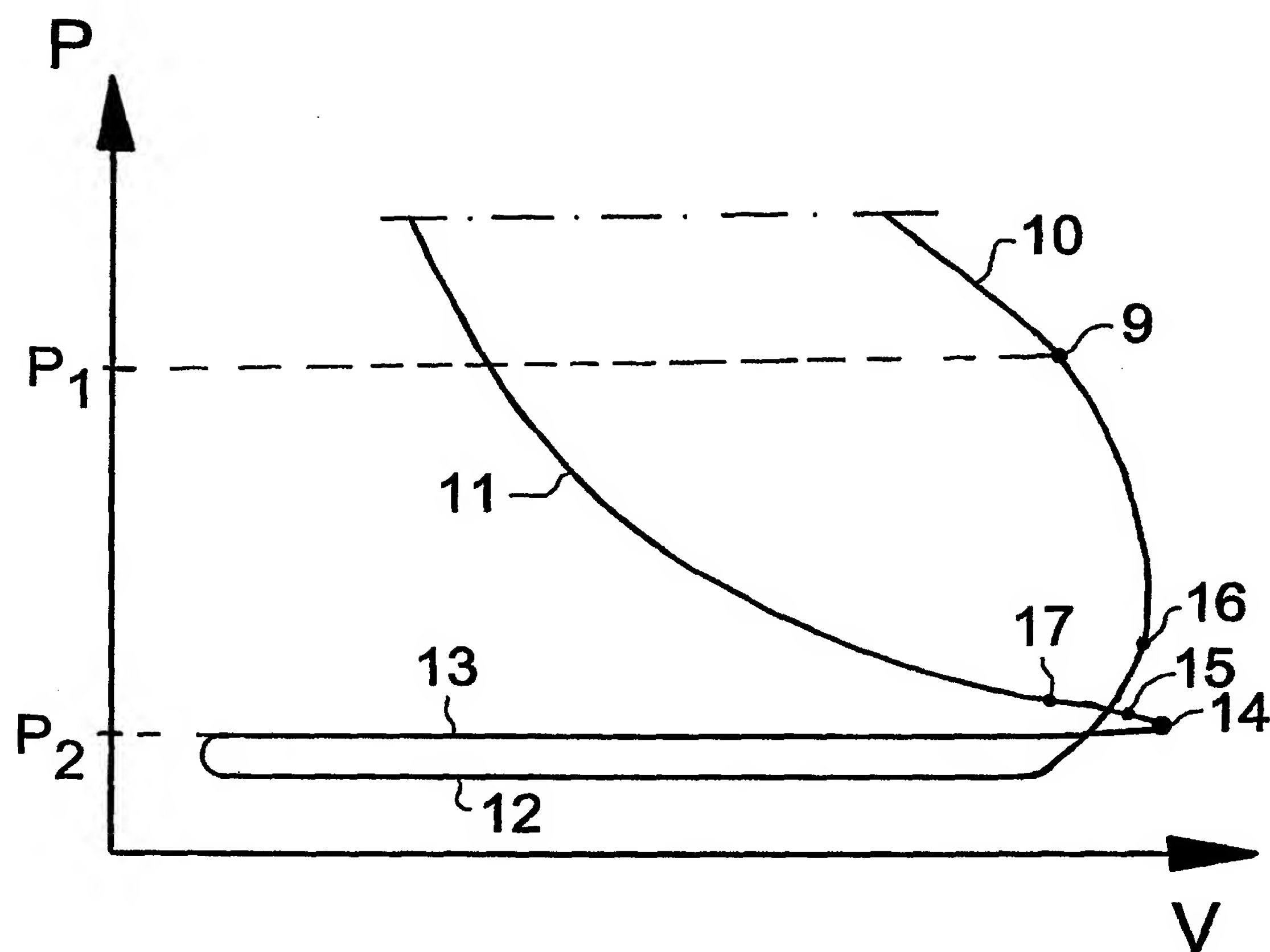


Fig 5

INTERNATIONAL SEARCH REPORT

International application No.

PCT/SE 98/02335

A. CLASSIFICATION OF SUBJECT MATTER

IPC6: F02M 25/07, F02B 47/08, F02D 21/08

According to International Patent Classification (IPC) or to both national classification and IPC

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C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	Derwent's abstract, No C3709 K/07, week 8307, ABSTRACT OF SU, 918467 (ALTAI POLY), 17 April 1982 (17.04.82), column 3, line 23 - line 25, abstract	1,4-8
Y	--	2,3
Y	EP 0235481 A1 (INSTITUT FRANCAIS DU PETROLE), 9 Sept 1987 (09.09.87), column 1, line 19 - line 32; column 4, line 1 - line 6, figure 2, abstract	2,3
A	US 4422430 A (WIATRAK), 27 December 1983 (27.12.83), abstract	
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INTERNATIONAL SEARCH REPORT

Information on patent family members

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Patent document cited in search report	Publication date	Patent family member(s)	Publication date
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		DE 3683036 A	30/01/92
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